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ABSTRACT

A computer program to analyze full-coverage film-cooled vanes and blades with or without a thermal-barrier coating is described. The program input, coolant flow and heat transfer model, and the program output are discussed. As an example, sections of the suction and pressure sides of a high temperature, high pressure turbine vane are analyzed to show the effects of a thermal barrier coating. Compared to the uncoated design, the coating halves the required coolant flow, while simultaneously reducing metal outer temperatures by over 111 K (200° F).

NOMENCLATURE

Symbols

a_1, a_2	constants in metal and coolant temperature expressions
c_p	specific heat at constant pressure, J/(g·K); Btu/(lbm·°R)
$C_2,\ldots C_6$	constants in metal and coolant temperature and overall effectiveness expressions
D	diameter, m; ft
G	flow rate per unit area, kg/(m^2·hr); lbm/(ft^2·hr)
g_c	force-mass conversion constant, 1; 32.174 (lbm)(ft)/(lbf)(sec^2)
h	heat-transfer coefficient, J/(m^2·hr·K); Btu/(ft^2·hr·°R)
$h_g(0,x)$	mainstream gas heat transfer coefficient when the coolant temperature equals the mainstream gas temperature, J/(m^2·hr·K); Btu/(ft^2·hr·°R)
$h_g(1,x)$	mainstream gas heat transfer coefficient when the coolant temperature equals the shell outer surface temperature, J/(m^2·hr·K); Btu/(ft^2·hr·°R)
k	thermal conductivity, J/(m·hr·K); Btu/(ft·hr·°R)
l	length, m; ft
p	pressure, N/m^2; lbf/ft^2
Pr	Prandtl number
R	gas constant, J/(kg·K); ft·lbf/(lbm·°R)
Re	Reynolds number
r	radial location, m; ft
T	temperature, K; °R
V	velocity, m/sec; ft/sec
x	distance, m; ft
α	angle defining film-cooling hole orientation, deg
α_1, α_2	constants in overall effectiveness expression
β	angle defining film-cooling hole orientation, deg
η	overall effectiveness
λ_1, λ_2	constants in metal and coolant temperature and overall effectiveness expression
ρ	density, kg/m^3; lbm/ft^3
ω	rotational speed, rad/sec

Subscripts

a	based on arrival velocity
av	average
b	bulk
c	coolant
ct	coating
d	based on diameter
g	mainstream gas
i	inner, inlet
if	interface
m	metal
n	based on impingement hole centers
o	outer, outlet
w	wall
0	reference location from engine center line
1, 2	arbitrary locations along the film-cooling hole length
∞	supply

INTRODUCTION

The high operating temperatures and pressures of modern jet engines require cooled vanes and blades to ensure the structural integrity of the turbine. Convection and/or local film cooling are being used in current engines, but any further significant increase in engine operating temperatures and pressures will require more effective cooling schemes to keep coolant flow rates at acceptable levels. Two such schemes which have been investigated by NASA are full-coverage film-cooling and the use of thermal barrier coatings. In full-coverage film-cooling, compressor discharge air is first impinged on a vane or blade inner surface to remove heat by convection and is then bled out through a large number of evenly distributed holes in the outer surface. The coolant forms a continuous, relatively cool instulating layer between the outer surface and the hot mainstream gas. A thermal barrier coating acts as a heat insulator which, when used with back side convection cooling, can lower metal temperatures significantly.

While numerous aspects of full-coverage film-cooling had been reported in the open literature (1-5), a computer program to assist in the design of such vanes and blades was not generally available. A computer program was therefore developed for analyzing full-coverage film-cooled vanes or blades, including the effects of a thermal barrier coating. The program is described in detail in reference 6.

This paper presents an overview of the computer program of reference 6 and illustrates its use with an example problem showing the effects of a thermal barrier coating. Sections of the suction and pressure sides of a high-temperature high pressure turbine vane are analyzed and the resulting coolant flows and metal temperatures are discussed.

ANALYSIS MODEL

Geometry

Figures 1 and 2 show a typical full-coverage film-cooled vane and its cross section, respectively. The cross section shows internal ribs which, together with an insert, divide a vane or blade into chambers. The large variations in mainstream gas pressure around the airfoil periphery make chambers necessary to properly control and meter the coolant flow. The analysis considers a single chamber in a vane or blade. An airfoil is designed by analyzing each chamber in the vane or blade.

Coolant Flow

The coolant flow path in a chamber is shown in figure 2. Coolant flows through the insert holes, impinges on the shell inner surface, and then flows out through film-cooling holes into the mainstream gas.

Coolant flow through each chamber is treated as one-dimensional and compressible. Flow through the impingement and film-cooling holes is treated in terms of a discharge coefficient and a total pressure loss coefficient (5), respectively. The coolant temperature increase due to impingement on the shell inner surface and convection in the film-cooling holes is calculated. For a rotating blade, the radial pressure distribution $p(r)$ in the chamber is assumed to be that of a rotating stagnant column

$$p(r) = p_0 \exp\left[\frac{\omega^2\left(r^2 - r_0^2\right)}{2RTg_c}\right]$$

where p_0 is a known pressure at the radius r_0. The program can be used for flow analysis only, by specifying the exclusion of heat transfer calculations.

Heat Transfer

Heat transfer calculations for a coated or uncoated shell are one-dimensional and are based on the equations of reference 2, which were modified to include a thermal barrier coating. Calculated shell temperatures are average values for an area of shell outer surface associated with each film-cooling hole row. The calculations for each film-cooling hole row include back-side impingement and convective heat transfer in the holes, but conduction between adjacent hole rows is neglected. All shell and coolant temperatures (defined in fig. 2) are determined in an iterative manner. For a coated shell, the coating outer temperature $T_{ct,o}$, metal outer and inner temperatures ($T_{m,o}$ and $T_{m,i}$, respectively) and the coolant outlet, interface and inlet temperatures ($T_{c,o}$, $T_{c,if}$ and $T_{c,i}$, respectively) are given by (6)

$$T_{ct,o} = T_g - \frac{(T_g - T_{c,\infty})[\eta G_c C_p + (1 - \eta)\Delta h_g]}{h_g(0,x) - \eta\,\Delta h_g + \eta G_c C_p}$$

$$T_{m,o} = (T_{ct,o} - T_{c,\infty})\left(C_2 e^{a_1} + C_3 e^{a_2}\right) + T_{c,\infty}$$

$$T_{m,i} = (T_{ct,o} - T_{c,\infty})(C_2 + C_3) + T_{c,\infty}$$

$$T_{c,o} = \eta(T_{ct,o} - T_{c,\infty}) + T_{c,\infty}$$

$$T_{c,if} = (T_{ct,o} - T_{c,\infty})\left[C_2\left(1 - \frac{a_1^2}{\lambda_1}\right)e^{a_1} + C_3\left(1 - \frac{a_2^2}{\lambda_1}\right)e^{a_2}\right] + T_{c,\infty}$$

$$T_{c,i} = (T_{ct,o} - T_{c,\infty})\left[C_2\left(1 - \frac{a_1^2}{\lambda_1}\right) + C_3\left(1 - \frac{a_2^2}{\lambda_1}\right)\right] + T_{c,\infty}$$

where the overall effectiveness η is obtained from

$$\eta = C_4 + C_5\left(1 - \frac{\alpha_1^2}{\lambda_2}\right)e^{\alpha_1} + C_6\left(1 - \frac{\alpha_2^2}{\lambda_2}\right)e^{\alpha_2}$$

and

$$\Delta h_g = h_g(0,x) - h_g(1,x)$$

The heat transfer coefficients $h_g(0,x)$ and $h_g(1,x)$ are obtained from reference 3 which was modified to include the discrete-hole blowing model of reference 4. The constants a_1, a_2, C_2, C_3, C_4, C_5, C_6, α_1, α_2, λ_1, and λ_2 are functions of geometry, physical properties, coolant flow rate, and impingement and film-cooling hole heat transfer coefficients. They are defined in reference 6.

The heat-transfer coefficient on the shell inner surface is calculated from the Gardon-Cobonpue impingement correlation (7)

$$h_{av} = \frac{0.286\ k(Re_{a,n})^{0.625}}{x_n}$$

The heat-transfer coefficient in the film-cooling holes is calculated from the Davey correlation (8), from which the average h in the portion of the hole between stations l_1 and l_2 is

$$h_{av} = \left\{0.045\left(\frac{k}{D}\right)(Re_d)^{0.8}(Pr)^{0.4}\left(\frac{T_b}{T_w}\right)^{0.18} \times D^{0.2}\left[(l_2)^{0.8} - (l_1)^{0.8}\right]\right\} \Big/ l_2 - l_1$$

COMPUTER PROGRAM

The computer program includes a main program and six subroutines. The program consists of 1650 cards and occupies 22,500 36-bit words of memory. It is written in FORTRAN IV and is operational on a UNIVAC 1100/42 computer. Execution time for one chamber is typically less than 15 seconds.

Input

For each chamber in a vane or blade, the number of impingement and film-cooling hole rows are specified, along with the number of holes in each row. Further required goemetry input is shown in figure 3. Each film-cooling hole row has an associated area of outer surface as shown. The film-cooling hole orientation is described by the two angles shown in figure 4. Additional input quantities are the coolant supply temperature and pressures, and mainstream gas side temperature, pressure, velocity and heat transfer coefficient distributions. The physical properties of the coolant and the thermal conductivities of the metal and ceramic coating are input as functions of temperature, and the coolant flow coefficients (obtained from ref. 5) as functions of Mach number. Units may be either SI or U.S. customary.

Output

For each chamber, the program output consists of a listing of the geometric variables and the calculated flow and heat transfer results. For each impingement hole row, the coolant pressure, temperature, Mach number, and coolant flow rate are tabulated. These variables are also tabulated for the entrance and exit stations of each film-cooling hole row. The pressure or pressure distribution (for a rotating blade) in the chamber is shown, along with the number of iterations required to achieve flow convergence. When heat transfer calculations are specified, the heat-transfer coefficients on the shell inner surface and in the film-cooling holes are tabulated for each film-cooling hole row, along with the metal inner and outer temperatures and the coating outer temperature. The number of iterations required for the heat-transfer calculations is also shown and appropriate error messages are printed out for all calculations.

EFFECTS OF A THERMAL BARRIER COATING

A single chamber was analyzed on each of the suction and pressure sides of a high-temperature high-pressure turbine vane (fig. 2) to illustrate the use of the computer program. The thermal barrier coating thickness and the coolant flow rate were varied independently and the metal shell inner and outer and coating outer temperatures were calculated. The vane material was MAR-M509 and the coating was 12 weight percent yttria stabilized zirconia (Y_2O_3-ZrO_2), plasma sprayed over a bond coat of NiCrAlY (Ni-16Cr-6Al-0.5Y). The combined thickness of the metal shell and the plasma sprayed bond coat was 0.127 cm (0.050 in.). The effects of outer coating thicknesses from 0.0127 to 0.076 cm (0.005 to 0.030 in.) were analyzed.

In varying the coolant flow rate the computer program was run repeatedly with varying coolant hole sizes which were smaller than those of an uncoated base-case design. The hole sizes were established in an iterative manner to maintain constant coolant-to-mainstream mass flux ratios $(\rho V)_c/(\rho V)_g$. Coolant flow reductions were limited to those attainable with 0.0254 cm (0.010 in.) diameter film-cooling holes. Film-cooling holes smaller than 0.0254 cm were excluded due to manufacturing considerations.

Assumptions

1. The operating conditions are a turbine inlet hot spot total temperature of 2550 K (4130° F) (corresponding to an average inlet temperature of 2200 K (3500° F)), and an inlet total pressure of 38 atmosphere (385 N/cm^2 or 559 psia). Coolant temperature and pressure are 811 K and 404 N/cm^2 (1000° F and 586 psia), respectively.

2. Thermal gas radiation is neglected in the analysis. For an uncoated vane at the chosen operating conditions, thermal gas radiation increases the heat flux by approximately 5 percent. For a coated vane, the heat flux increase is approximately 1 percent due to the higher reflectance of the zirconia, which is 0.8 as compared to 0.2 for the base metal. Neglecting gas radiation thus results in a conservative estimate for the benefits of a thermal barrier coating.

3. The coating has been polished smooth so that mainstream gas heat-transfer coefficients do not increase due to surface roughness.

4. The boundary layers on the vane suction and pressure sides are fully turbulent.

5. Temperature calculations with ceramic coating are made with a two-layer model (metal plus coating). The thin bond layer is combined with the base metal since the thermal conductivity of each is much greater than that of the thermal barrier coating. This model introduces little error into the analysis.

Results

Figures 5 through 7 present the calculated results. The shown temperatures are average values for all film-cooling hole rows in the chosen chambers. The results are compared to the uncoated base-case vane design for which the average suction and pressure side chamber outer metal temperatures are 1328 and 1322 K (1930° and 1920° F), respectively.

Figures 5(a) and (b) show the coating outer temperature ($T_{ct,o}$), the metal outer temperature ($T_{m,o}$), and the metal inner temperature ($T_{m,i}$) at the uncoated design coolant flow versus coating thickness. Also shown are the uncoated design metal outer temperatures. Even small coating thicknesses reduce the metal outer temperatures significantly. For example, a 0.0127 cm (0.005 in.) coating reduces the metal outer temperature by 189 and 150 K (340° and 270° F) for the suction and pressure side chambers, respectively.

Figures 6(a) and (b) show the coating outer temperature ($T_{ct,o}$) and the metal outer temperature ($T_{m,o}$) for various coating thicknesses versus coolant flow for the suction and pressure side chambers, respectively. These figures show that a thermal barrier coating allows significant reductions in metal temperature, coolant flow, or combinations of both, and that metal outer temperatures are more sensitive to coating thickness than to coolant flow. For example, with a 0.0127 cm (0.005 in.) coating and with half the uncoated design coolant flow, suction and pressure side metal outer temperatures are reduced 133 and 83 K (240° and 150° F) respectively, compared to the uncoated design. At twice this coolant flow, these reductions are 189 and 150 K (340° and 270° F), respectively. Doubling the coating thickness from 0.0127 to 0.0254 cm (0.005 to 0.010 in.) (at half the uncoated design coolant flow) reduces the suction and pressure side metal outer temperatures by 228 and 178 K (410° and 320° F) respectively, compared to the uncoated design.

Figures 7(a) and (b) (obtained by cross plotting figs. 6(a) and (b)) show the required coating thicknesses for the suction and pressure side chambers, respectively, versus metal outer temperature at various coolant flow rates. These figures show that, to achieve the same metal outer temperature reductions on the suction and pressure sides at equal coolant flow reductions, a thicker coating is required on the pressure side than on the suction side. For example, at half the uncoated design coolant flow, a 0.0127 cm (0.005 in.) thick coating reduces the suction side metal outer temperature (compared to the uncoated design) by 133 K (240° F) to 1194 K (1690° F), while a 0.0178 cm (0.007 in.) thick coating is required for a comparable temperature reduction (to 1189 K (1680° F)) on the pressure side. The greater effectiveness of the thermal barrier coating on the suction side results from the higher heat flux on the uncoated suction side (ref. 9 points out that the benefits of a thermal barrier coating are directly related to the level of heat flux through the uncoated hardware).

CONCLUDING REMARKS

An overview has been presented of a computer program which analyzes the coolant flow and metal and coating temperatures of a chamber in a full-coverage film-cooled vane or blade. The program geometric inputs are shown, and other program inputs are described, as well as the program output. The basic equations for calculating metal and coolant temperatures are shown and various heat-transfer correlations are described. As an example, sections on the suction and pressure sides of a high-temperature high-pressure turbine vane are analyzed to show the effects of a thermal barrier coating. The analysis shows that combining a thermal barrier coating with full-coverage film-cooling gives significant reductions in metal temperature and/or coolant flow rates. For the chosen vane sections, required coolant flow rates were halved (compared to an uncoated design), while metal outer temperatures were simultaneously reduced by over 111 K (200° F). While NASA has demonstrated the use of plasma-sprayed thermal-barrier coating on convection-cooled vanes (9), research is needed to develop an acceptable manufacturing technique for coated, full-coverage-film-cooled hardware.

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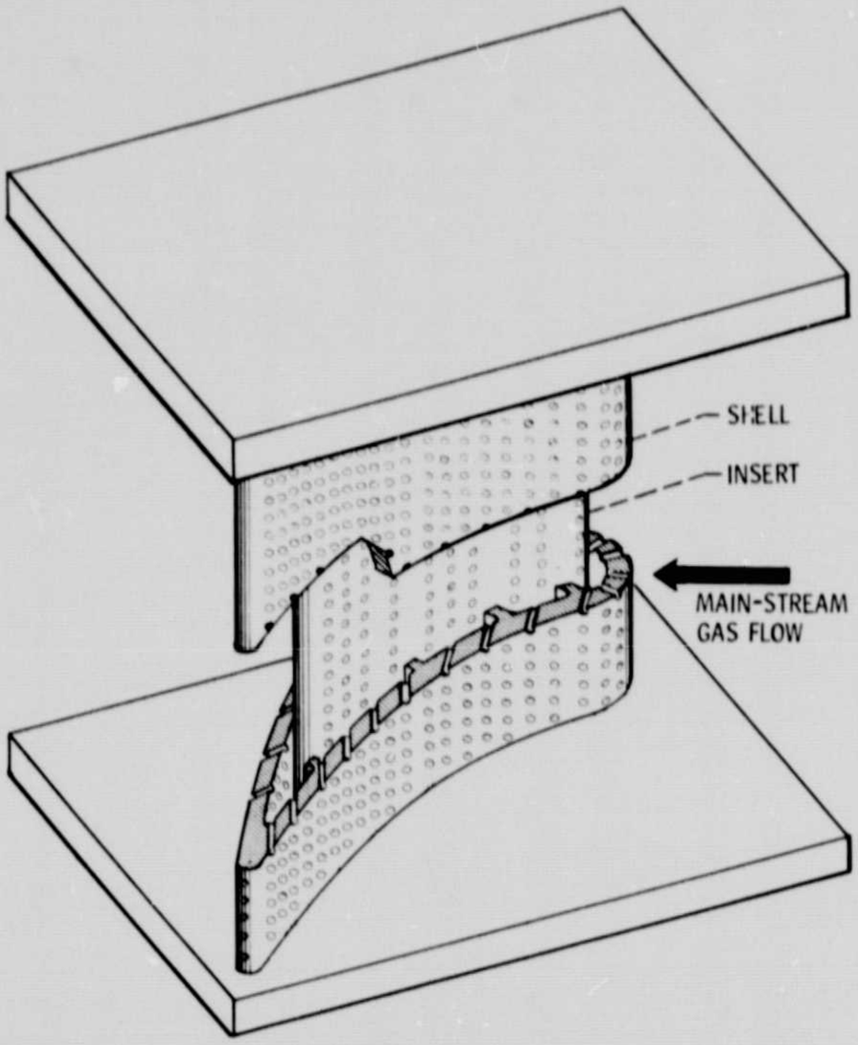

Fig. 1 Full-coverage film-cooled vane

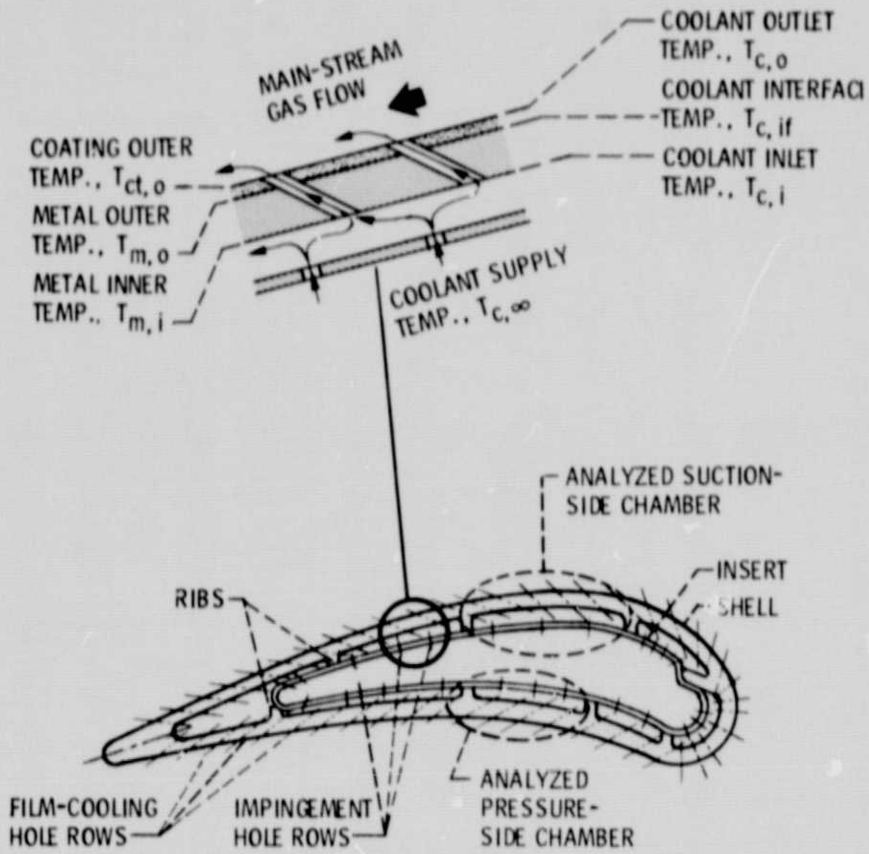

Fig. 2 Full-coverage film-cooled vane cross-section and temperature definitions

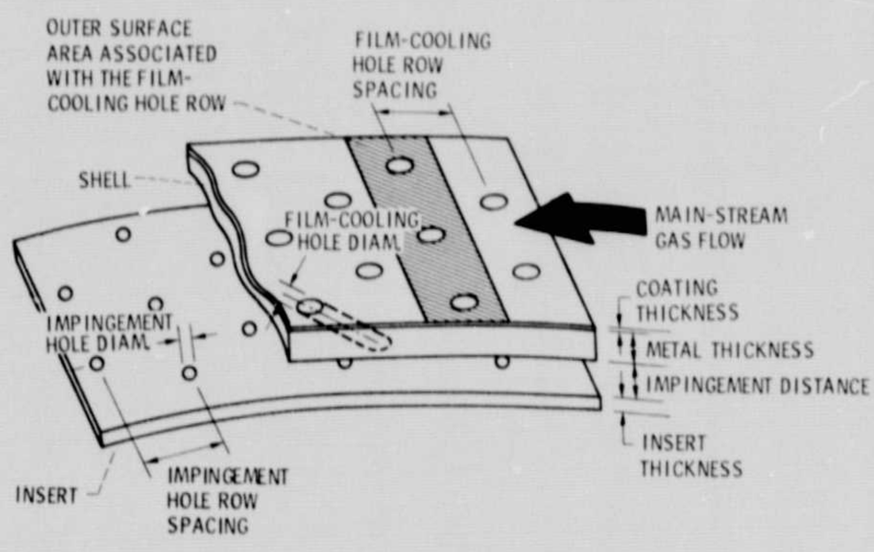

Fig. 3 Computer program geometry input

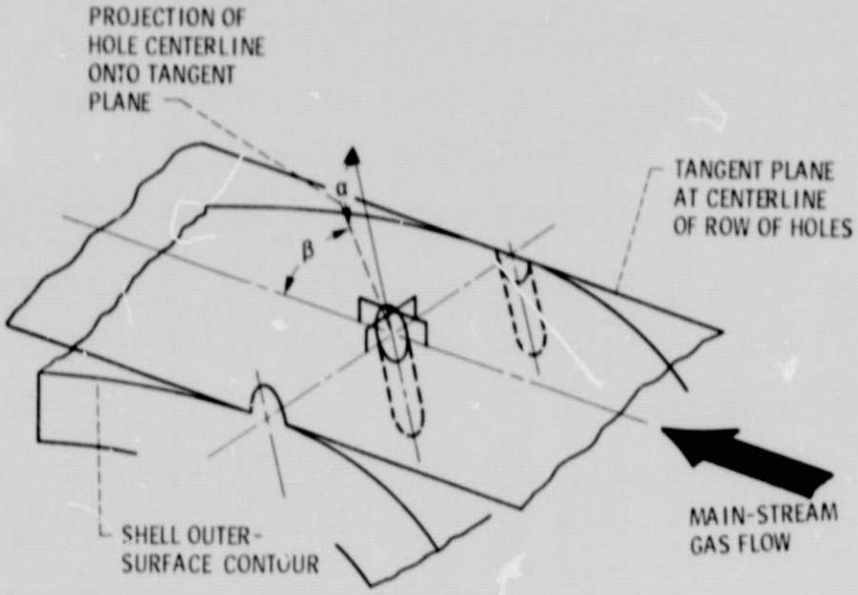

Fig. 4 Film-cooling hole orientation

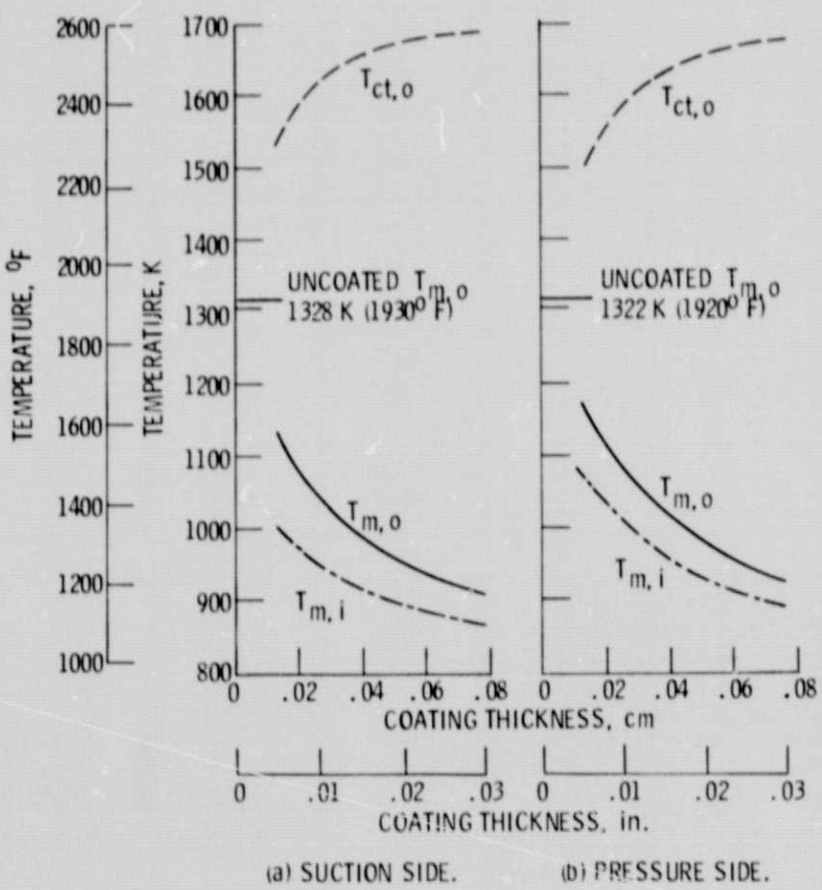

Fig. 5 Variation of chamber average temperature with coating thickness at uncoated design coolant flow

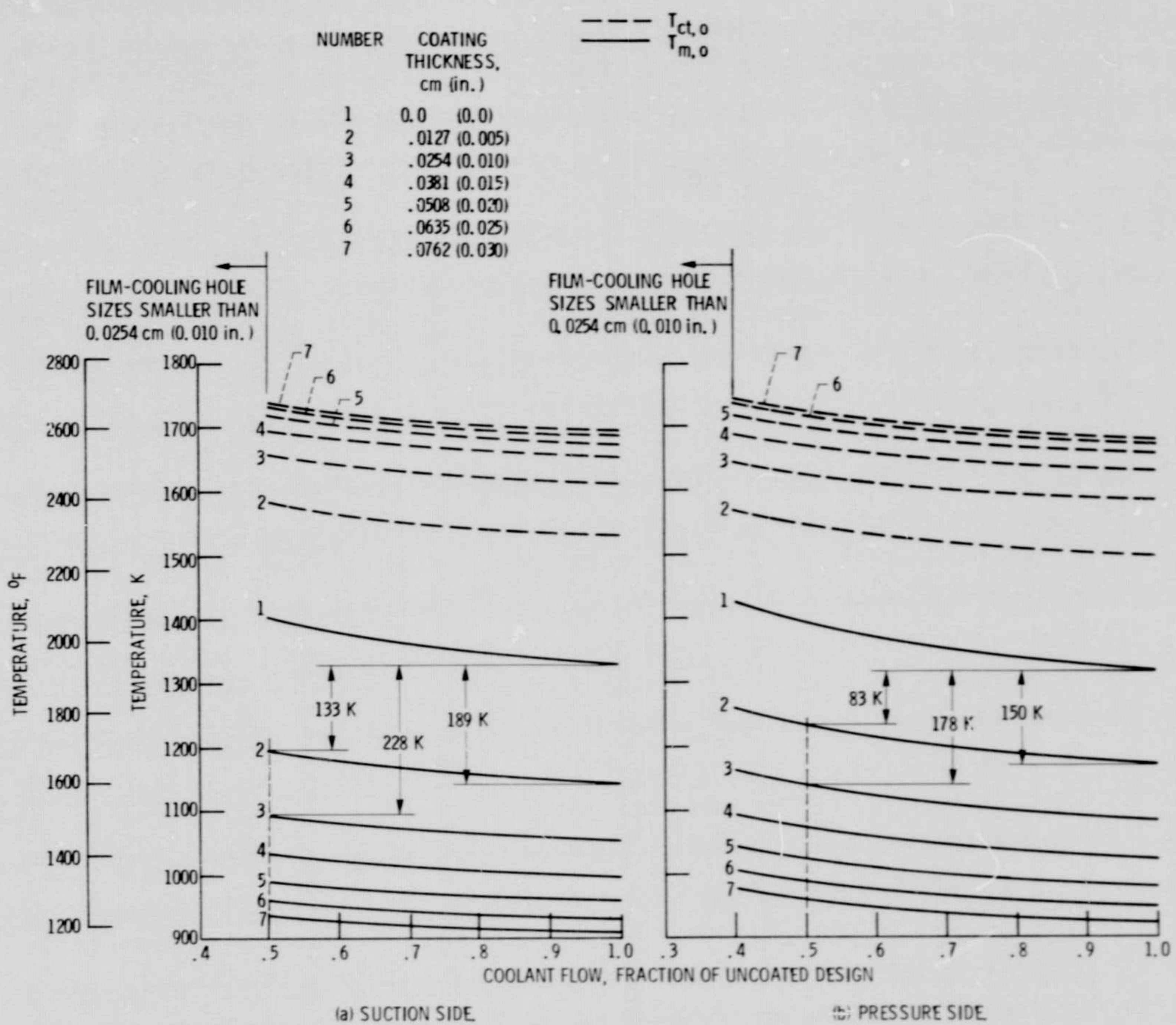

Fig. 6 Variation of chamber average temperatures with coolant flow at various coating thicknesses

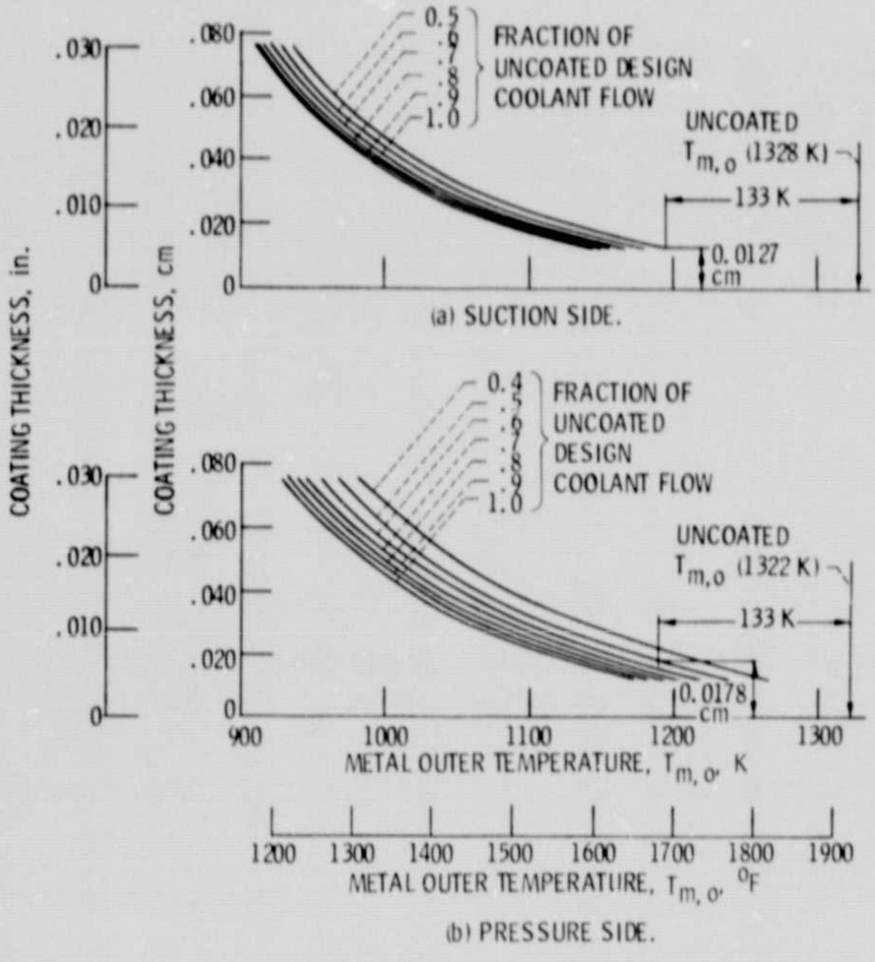

Fig. 7 Variation of coating thickness with chamber average metal outer temperature at various coolant flows

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16. Abstract
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